3.E.2.f(4)

(b) The fluids may oxidize when exposed to air. Sensitive temperature control is required to prevent boiling, coking, scaling, or fluid deterioration due to high fluid film temperatures in the heater.

(c) Deterioration in service can result in lowering of the flash point and increase in carbon residue.

(d) The fluid should be handled cautiously since exposure to skin, eyes, lungs, or the digestive system can be irritating or cause illness.

(e) Heat transfer fluids have a lower coefficient of heat transfer on the inside tube surface than does steam, necessitating higher operating temperatures or more surface area for the same performance. Fluid circulating pumps can be high maintenance items.

(f) Also, although there are some offsetting factors in the cost of these fluids, they are expensive.

Work is being done to develop water-based heat transfer fluids to reduce costs. Water has some advantages over oil in that it has excellent heat transfer properties, low viscosity, low vapor pressure, high thermal conductivity, and high specific heat and density. It is not combustible or toxic. Additives are being developed to reduce its limitations regarding operating temperature, corrosion, freezing, etc. This use of water is promising but has had little success so far. [NOTE: The preceding discussion is intended as an introduction to TFHs used in marine applications. Large heaters found in the chemical processing industry are normally much more sophisticated in design.]

g. Pressure Vessels.

(1) General Requirements.

(a) Historical Basis. In the early 20th Century, materials and technology were such that pressure vessels were limited to a capacity of a few hundred pounds per square inch (psi) pressure. Even at that, explosions of such equipment were not uncommon. A real need was realized for the development of materials, design and fabrication methods, protective devices, and quality control procedures. Cities, states, and other jurisdictions began development of design, construction, and inspection rules to help prevent pressure vessel failures. As these rules were enforced, it became more and more difficult to construct a pressure vessel in one jurisdiction that would be accepted in another, due to conflicts in the applicable rules. Because of this lack of uniformity, the ASME Pressure Vessel Code was developed in 1925. This was an attempt to offer the various jurisdictions a standard set of design and construction rules for safe pressure vessels. Much progress has occurred since that time. Pressure vessels are now being built for service at pressures of several thousand psig and equally severe

3.E.2.g(1)

(a) (cont'd) temperatures. The Code is continuously updated by the ASME Boiler and Pressure Vessel Committee to keep up with growth in the pressure vessel industry.

(b) Coast Guard Adoption of the ASME Code. On 1 July 1969, the Coast Guard adopted the ASME Code for pressure vessels under its jurisdiction. Section VIII, Division 1 is adopted for most, but not all, pressure vessels aboard Coast Guard certificated vessels. 46 CFR 54 adopts the code, while modifying some of its provisions to conform more appropriately to marine practice. Therefore, a comprehensive understanding of pressure vessel requirements can only be achieved by reading the ASME Code, along with the modifications from Part 54. The code applies to many types of welded and forged pressure vessels. However, there are many pressure vessels outside the scope of the code, as adopted, that must be reviewed on an individual basis. These include non-metallic vessels, wire-wound vessels, multi-layer vessels with shrink-fitted shells, vessels with design pressures in excess of 3000 psig, etc. These types of pressure vessels are normally submitted to Commandant (G-MSE-3) for review. There have been few marine applications of these vessels to date; however, the use of very high-pressure accumulators for hydropneumatic service is becoming more common.

(c) Exemptions From Compliance. 46 CFR Part 54 exempts certain categories of pressure vessels from plan review and shop inspection for various reasons. Some exemptions are conditional on the presence of the "U" or "UN" code symbol. The "U" stamp signifies that the pressure vessel complies with the applicable design, fabrication, and testing requirements of the code, and has been inspected by an authorized inspector. The "UN" stamp indicates code compliance, except that the independent third party inspection is not required (see paragraph U-1 of Section VIII, Division 1 for applicability of the "UM" stamp). The presence of a code stamp is not required to be on pressure vessels that must receive Coast Guard plan approval and shop inspection. The Coast Guard issued Final Rules, effective 21 June 1982, that require Class I, II, and III pressure vessels not containing hazardous materials to be inspected and stamped in accordance with the ASME Code. These rules replace previous requirements for plan approval and shop inspection by the Coast Guard. These rules require:

i. Certification of pressure vessel design drawings and analyses by a registered professional engineer (under 54.01-5(e));

ii. Coast Guard inspection of the completed pressure vessels prior to installation, 54.10-3(c); and

iii. Compliance with certain requirements which are optional under the ASME Code (see 54.01-5(d)).

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(d) Revisions of the ASME Code. The Boiler and Pressure Vessel Committee meets regularly to consider proposed additions and revisions to the code and to formulate code cases. Proposed changes are published in ASME's magazine Mechanical Engineering for public comment from all interested persons. After the allotted time for public review and final acceptance by the ASME Council, the Addenda (which include the additions and revisions to individual sections of the code) are published twice a year. These addenda are accepted by the Coast Guard for new construction 6 months after their date of issue. Code cases do not revise the code. They are issued to clarify the intent of existing requirements or provide, when the need is urgent, rules for materials or construction not covered by existing Code rules. Code cases are published in two separate books, Boilers and Pressure Vessels and Nuclear Components. Code cases are not adopted by the Coast Guard unless specifically authorized by the Commandant.

(e) Coast Guard Modifications of the ASME Code.

i. One of the Coast Guard modifications to the code is the division of pressure vessels into various classes such as Class I, Class II-L, etc. These categories are used in defining exemptions from Coast Guard requirements, or in specifying additional requirements, such as for welded joints, nondestructive testing, or heat treatment (see 46 CFR Table 54.01-5(b)). These additional requirements provide for specific marine service and, in general, represent owner-option alternatives under the code. The jurisdiction of the code with regard to external piping ends at the first circumferential joint in welded end connections, the face of the first flange in bolted-flange connections, and the first threaded joint in threaded pipe connections.

ii. The requirements that 46 CFR 54 contain "modifications" to the adopted code are significant and straightforward, and should be reviewed by marine safety personnel approving or inspecting pressure vessels. They discuss loadings due to ship's motions, corrosion allowance and protection, external pressure, low temperature service, toughness testing of materials, inspections, stamping, data reports, pressure relief devices, pressure relief under fire conditions, welding, nondestructive examination, materials, and stress relief.

(2) Stress Calculation.

(a) Introduction.

i. To determine the allowable design stresses for multi-axial stress conditions that occur in pressure vessels, several theories of failure have been developed. Since all sections of the ASME Code do not base design requirements on the same failure theories, it is worthwhile to briefly consider these concepts. The

3.E.2.g(2)(a)

i. (cont'd) purpose of the failure theories is to predict when failure will occur due to combined stresses on the basis of data gained from simple uniaxial tests.

ii. Section I and Section VIII, Division 1 are based on the "Maximum Stress" or "Rankine" Theory. Under this theory, failure occurs when one of the principal stresses reaches the yield point value in tension or compression. The principal stresses in a cylinder are axial stress, radial stress, and tangential or hoop stress. Section III (Nuclear Power) and Section VIII, Division 2 are based on the "Maximum Shear Stress" or "Tresca" Theory. Under this theory, failure occurs when the maximum shear stress reaches some critical value, which depends on what type of loading the pressure vessel part is experiencing. The equation for stress analysis known as the Lame Equation, which is frequently used in analysis of thick wall cylinders, is based on the maximum stress theory. The ASME modified membrane equation,

$$t = \frac{PR}{SE - 0.6P} + C$$

is in very close agreement with the Lame Equation, and also takes into account weld joint efficiency and corrosion allowance (see ASME Section VIII, Division 1, UG-27 for definition of symbols). In a thin cylinder under internal pressure, where the radial stress is close to zero, the maximum stress and maximum shear stress theories give approximately the same results. However, in thick-wall cylinders having a radial stress that is not small in comparison to axial and hoop stress, the maximum shear stress theory will give results that coincide more closely with experimental data. It should not be concluded, however, that a thin-wall pressure vessel designed under Section VIII, Division 1 will be the same as one designed per Section VIII, Division 2. Safety factors and other parameters in the code sections differ significantly, as will be discussed in more detail later. One important characteristic of the maximum shear stress theory is that it can predict the failure of a material under either static or fatigue loading with good accuracy. It also gives excellent agreement with experimental results in the case of high tensile steels.

(b) <u>Alternate Stress Theories</u>. Two other failure theories should be mentioned, although they are not widely used in pressure vessel design. In the "Maximum Strain Energy" Theory (also known as the "distortion energy" or "Von Mises" Theory), rupture occurs when the strain energy per unit volume reaches a critical value. In the "maximum strain" theory, the part will fail when the maximum strain equals the strain at the elastic limit under simple tension.

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(c) Application of Engineering Judgment. One cannot be satisfied with a determination of stresses from a design analysis until all loads have been adequately considered. Paragraph UG-22 of Section VIII, Division 1 lists several types of loading that may be imposed on a pressure vessel, and that should be taken into account by the designer. 46 CFR 54.01-30 adds to this list by including static and dynamic factors peculiar to marine applications. During plan approval, some engineering judgment must be applied in deciding which of these loads will have a potentially significant effect on vessel stresses. For example, large low pressure tanks containing liquids may be highly stressed by both static liquid head and saddle supports. Liquid contents raise the reaction forces of the saddles. Sloshing loads could also be high in large liquid-containing vessels when baffles are not provided. Changing orientation of a ship (list and trim) or dynamic characteristics of a particular ship may sometimes need to be determined and used in defining pressure vessel loads. In most cases when a ship's motions should be considered, a simplified method that has proven satisfactory in the past has been to apply a static weight factor of 2G downward, and a 1G force in the fore-and-aft and athwartships directions. For the vast majority of pressure vessels approved under Part 54, dynamic loading from ship's motions need not be checked. Good engineering judgment must be applied in every case.

(d) Design of Saddle Supports for Horizontal Pressure Vessels. This is usually done with the assistance of a paper written by L. P. Zick, although other methods (including finite element analysis) may be used. Zick's paper, "Stresses in Large Horizontal Cylindrical Pressure Vessels on Two Saddle Supports," was published in the Welding Journal, Res. Suppl. 30 (1951). Although Zick's paper may be modified to accommodate vessels on three saddle supports, use of more than two saddles should usually be discouraged on a flexible foundation such as a ship. If all supports are not perfectly aligned, additional loads will be imposed on the vessel. Although Paragraph UG-22 requires that loadings must be considered, it does not give clear information on how to do so.

(e) Additional References. Some other references that may be useful in approving pressure vessel plans are:

i. Process Equipment Design, by Brownell and Young; John Wiley and Sons, Inc.

ii. Pressure Vessels - The ASME Code Simplified, by Chuse; McGraw-Hill Book Company, Inc.

iii. Theory and Design of Modern Pressure Vessels, by Harvey Van Nostrand; Reinhold Publishing Company.

iv. Defects and Failures in Pressure Vessels and Piping, by Thielsch, Reinhold Publishing Company.

3.E.2.g(2)(e)

v. Formulas for Stress and Strain, by Roark; McGraw-Hill Book Company, Inc.

(3) ASME Code, Section VIII, Division 1.

(a) Introduction. Since Section VIII, Division 1 of the ASME Code is so frequently used, this section is provided to detail these rules and their correct application. Most comments are intended to clarify or explain the basis of code requirements. Much of what is said will apply to Section I of the Code as well. The basic design criteria of Section VIII, Division 1 of the ASME Code is to provide adequate wall thickness in a vessel so that the maximum membrane stress does not exceed the allowable stress. The maximum allowable tensile stress values permitted for the various materials are given in Subsection C, Table UCS-23 of the code. The basis for establishing the allowable stress values is given in Appendix P. At temperatures below the creep range, except for bolting materials, the allowable stresses are generally based on the lowest value of the following:

i. 1/4 of the specified minimum tensile strength at room temperature;

ii. 1/4 of the tensile strength at temperature;

iii. 5/8 of the specified minimum yield strength at room temperature for ferrous materials;

iv. 5/8 of the yield strength at temperature for ferrous materials;

v. 2/3 of the specified minimum yield strength at room temperature for nonferrous materials; or

vi. 2/3 of the yield strength at temperature for nonferrous materials.

The code provides for increasing these allowable stresses for certain austenitic and nonferrous material. These higher allowable values are not recommended for flanges and other strain sensitive uses.

(b) Temperatures Below The Creep Limit. If bolting materials are used at temperatures below the creep range, with material strength increased by heat treatment or strain hardening, the following additional limits apply:

i. 1/5 of the specified minimum tensile strength at room temperature;

ii. 1/4 of the specified minimum yield strength at room temperature;

iii. At temperatures above the creep range, the allowable tensile stresses are based on the lowest value of the following:

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a. 100 percent of the average stress for a creep rate of 0.01 percent in 1000 hours;
b. 80 percent of the minimum stress for rupture at the end of 100,000 hours; and
c. 67 percent of the average stress for rupture at the end of 100,000 hours.

The allowable compressive stress is the same as the allowable tensile stress value, or the value determined according to UG-23(b), whichever is lower.

(c) Localized and Secondary Bending Stresses. It is recognized that high localized and secondary bending stresses exist in code vessels. Design rules for construction details have been written to hold such stresses at a safe level consistent with experience. One reason for the 3,000 psig limit on the scope of Section VIII, Division 1, is that many of these construction details are not appropriate for higher pressure applications. The basic design of specific parts such as heads and shells are covered by code design rules. However, if a vessel is subjected to severe cyclic operation, is in some other severe service, or has a complex geometry not covered by the rules, additional stress analysis will probably be necessary. Since Division 1 is primarily for membrane vessels, stress in the radial direction is usually not considered.

(d) Hoop Stresses. The general design formulas are given in UG-27. The hoop stress formulas are limited to a wall thickness not exceeding one-half of the inside radius and pressure not exceeding 0.385 SE (symbols defined in the code). When these limits are exceeded, the UA-2 (A)(1)requirements must be followed. The longitudinal stress formulas are limited to a maximum thickness of one-half of the inside radius and a thickness not exceeding 1.25 SE, with UA-2 (A)(2) followed beyond these limits. In a thin-walled cylinder with hemispherical heads, the average hoop stress in the shell is about twice the average axial (longitudinal) stress in the shell and twice the average stress in any direction of the hemispherical head. The average radial stress on the cylinder and head is compressive and equals one-half the internal pressure. This shows that, for internal pressure, the average hoop stress usually controls. Design formulas for other than hemispherical heads are based on a combination of analytical stress analysis, experimental stress analysis, and experience. The flat head formula must be adjusted for various details of joint design.

(e) Failure Criteria. The failure criteria used for external pressure or axial compression is elastic instability (buckling) and yielding from compressive stress. Provisions are included for design of stiffening rings. The minimum required thickness is found by a trial-and-error method. Once again, some materials have lower temperature limits in the stress curves for external pressure than the same materials in the tensile stress tables.

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(f) Shell and Head Openings. Openings in shells and heads must be designed per UG-36 through UG-42. When this is not practical, the ligament efficiency rules of UG-53 may be used. In viewing a cross section of the opening, adequate excess material must exist in the vessel and nozzle wall, or be added around the opening, to replace the material missing in the corroded hole in the vessel wall. Appendix L gives examples of application of these rules that are very helpful. In addition to providing the area of reinforcement, adequate welds must be provided to attach the reinforcement metal, and the induced stresses must be evaluated. The goal is to compensate for the weakening effect of the opening with metal of a suitable profile so as not to introduce an overriding stress concentration itself.

(g) Welded Joints. In applying code requirements for welded joints, it must be remembered that they are minimum requirements; design loads may require construction that is more restrictive. Requirements for weld geometrics, sizes, and details are contained in Part 13W. Section VIII, Division 1, as it is presently written, generally bases allowable stresses on one-fourth of ultimate tensile strength, as discussed earlier. However, to allow this stress value and the associated wall thickness and safety factors, the code requires mandatory examination of all butt welds by radiography. When butt welds are not radiographed or where butt welds are not used, the wall thickness and safety factor required are increased by Factor E in the design formulas. Factor E is referred to as the "joint efficiency" of the weld. In fact, this terminology is not all that appropriate. It is a carryover from before 1930, when most pressure vessels were of riveted construction. "Joint quality factor" is a more appropriate term, but "joint efficiency" will be used for consistency with the code. The intent of the code is to have three quality levels, one where all butt welds in the vessel are fully or partially radiographed, one for spot radiography, and one for welds without radiographic examination.

(h) Code Distinctions Covering Welded Joints. It is important, when doing pressure vessel plan review for the Coast Guard, that one understand the purposes and differences between the following paragraphs:

i. UW-2 Service Restrictions;
ii. UW-3 Welded Joint Categories;
iii. UW-11 Radiographic Examination;
iv. UW-12 Joint Efficiencies; and
v. Table UW-12, Maximum Allowable Joint Efficiencies.

A reviewer must have a complete understanding of service restrictions, joint design, and joint examination requirements to be able to correctly apply the code. This understanding is best gained by careful study of the content and intent of the code rules. If the design formula used has an E value selected from Table UW-12, the quality factor is

3.E.2.g(3)

(h) (cont'd) built in and no further consideration is required. However, formulas that do not contain an E value, which are used to calculate parts of vessels that include non-examined butt welds, require addition of an 80 percent quality factor or an 85 percent factor where spot radiography is used. Joint "category" designations are locations of joints in a vessel, and have no bearing on the type of joint. The February 1975 issue of ASME's Journal of Pressure Vessel Technology contains an article by G. M. Eisenberg that is of great assistance to understanding E factors and stress multipliers, and how they relate to requirements for radiography of butt welds; examples shown in Appendix "L" of the code will also be helpful. Since the degree of radiography affects the safety factor of a vessel designed to code rules, it helps to be reminded that no amount of radiography increases the strength of a weld. Only the assurance of weld quality is increased. [NOTE: Supplementary design rules are contained in Mandatory and Non-Mandatory Appendices of the code. Space does not permit discussion of all these provisions. However, those pages contain a wealth of requirements, recommendations, and information of which a code user should be aware.]

(4) ASME Code, Section VIII, Division 2.

(a) Introduction. Section VIII, Division 2 of the code contains alternative rules for pressure vessel design. Under Division 2, it is possible to design pressure vessels with a theoretical design margin (factor of safety) of three, based on the ultimate tensile strength of the material. This differs from Division 1, which essentially requires a design margin of five (which can be reduced to as low as 3.5 if certain procedures such as radiography are used).

These rules may not be used for portable pressure vessels other than pressure vessels for human occupancy (PVHO's) used in diving operations. Because of the demand for higher pressures and temperatures for pressure vessels and limitations on availability of materials, Division 2 was developed and published in 1968. Its main goal was to provide for better utilization of existing materials. What does this lower factor of safety really mean? Essentially, with the higher stresses allowed, it offers the possibility of manufacturing a lighter vessel, with thinner shells and heads and less weld metal; but the lighter vessel may carry a higher price tag. There are many restrictions on material selection; a very detailed stress analysis is required. Inspection and testing procedures are much more comprehensive than for Section VIII, Division 1 vessels. In fact, the costs of the extra engineering and inspection efforts will often outweigh the savings in material and fabrication costs.

(b) Professional Certification. One important feature of Division 2 is that the user of the vessel is required to present the manufacturer detailed information about intended operating conditions. This information must be certified by a registered professional engineer experienced in pressure